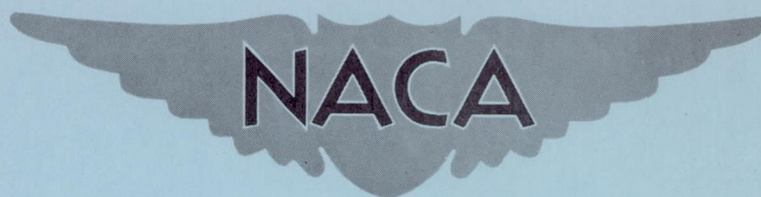


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# RESEARCH MEMORANDUM

INVESTIGATION OF CENTRIFUGAL COMPRESSOR OPERATED AS  
A CENTRIPETAL REFRIGERATION TURBINE

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NATIONAL ADVISORY COMMITTEE  
FOR AERONAUTICS

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## SUMMARY

A centrifugal compressor from a production-type turbojet engine was successfully operated as a centripetal refrigeration turbine over a range of rotor speed, inlet pressure, and pressure ratio for full admission; with a constant inlet pressure, the unit was operated over a range of rotor speed and pressure ratio for several degrees of partial admission. Performance data are presented for a range of nominal equivalent rotor speeds and pressure ratios. At a nominal equivalent rotor speed of 6800 rpm, performance results are presented for various turbine-inlet pressures (corresponding to percentages of throttling at the turbine inlet) and several different degrees of partial admission.

From the foregoing performance results, partial admission was concluded to be the more efficient method of turbine control, although throttling at the turbine inlet was easier to apply.

A study of the possible cause of erosion of the nozzles and the rotor observed during the investigation indicated that the nozzles should have erosion-resistant surfaces to minimize this detrimental effect.

## INTRODUCTION

An investigation was undertaken at the NACA Lewis laboratory to determine whether a centrifugal-flow compressor from a production-type turbojet engine could serve as an air-refrigeration unit in the laboratory service facilities when operated as a radial centripetal turbine. The over-all turbine performance was investigated in terms of air-temperature drop and adiabatic temperature-drop efficiency over a range of rotor speeds, inlet-air pressures, pressure ratios (corresponding to percentages of throttling at the turbine inlet), and degrees of partial admission. It was desired to establish the best turbine operating speed

for maximum refrigeration effect, to evaluate partial admission and inlet throttling as a possible means for controlling the air-temperature drop and the air-weight flow through the unit, and to obtain information on the practical aspects of operation of the unit.

A turbojet-engine centrifugal-flow compressor was therefore set up and the air flow through the unit was reversed so that it operated as a centripetal refrigeration turbine. Performance data are presented in terms of the variation in refrigerating effect (in Btu/sec), air horsepower, temperature drop, weight flow, and adiabatic efficiency with various over-all pressure ratios for both inlet throttling and partial admission.

An observed erosion problem is discussed and a possible method for minimizing its detrimental effect is recommended.

### SYMBOLS

The following symbols are used in this report:

$c_p$	specific heat at constant pressure, (Btu/(lb)(°R))
hp	horsepower, $778 w c_p \Delta T / 550 \delta \sqrt{\theta}$
N	rotor speed, rpm
$N/\sqrt{\theta}$	equivalent turbine-rotor speed, rpm
$p'$	total pressure, (in. Hg absolute)
Q	equivalent total heat extraction, (Btu/sec)
w	air-weight flow, (lb/sec)
$w\sqrt{\theta}/\delta$	equivalent air-weight flow, (lb/sec)
$\Delta T$	temperature drop across turbine, °R
$\Delta T/\theta$	equivalent temperature drop across turbine, °R
$\delta$	ratio of inlet-air pressure to NACA standard sea-level pressure (29.92 in. Hg absolute)
$\eta_t$	adiabatic temperature-drop efficiency
$\theta$	ratio of inlet-air temperature to NACA standard sea-level temperature (518.4° R)



## Subscripts:

- 1 inlet measuring station, upstream of inlet throttle
- 2 inlet measuring station, downstream of inlet throttle
- 3 outlet measuring station

## APPARATUS

A standard turbojet-engine centrifugal-compressor assembly consisting of a double-entry impeller, a vaned diffuser, and a compressor casing was used. When the normal air-flow direction through the compressor was reversed in this investigation, the unit became a turbine, the impeller became the rotor, the diffuser became the nozzles, and the compressor casing became the turbine casing.

The turbine nozzles (fig. 1(a)) were fabricated from a magnesium alloy and consisted of 14 individual passages, each with a throat area of 5.5 square inches. The nozzle-outlet angle, as shown on figure 1(a), was  $16\frac{10}{2}^\circ$ . The clearance between the trailing edges of the nozzle blades and the rotor blade tips was 2 inches.

The turbine rotor, fabricated from an aluminum alloy, was a double-entry type consisting of 31 blades per side (fig. 1(b)). The unit investigated was assembled without the normal accessory drives and mounted on a bulkhead which, in turn, was bolted to one end of an airtight steel depression tank 6 feet in diameter and approximately  $13\frac{1}{2}$  feet in length. The turbine rotor was coupled by a splined shaft to a 6.735:1 speed decreaser and was connected to a 9000-horsepower variable-frequency induction dynamometer that absorbed the turbine-power output.

The experimental setup of the refrigeration turbine is diagrammatically shown in figure 2. Air flow to the turbine (supplied by the laboratory refrigerated-air system) passed through a submerged adjustable orifice in the inlet ducting and an electronic precipitator, which removed foreign particles from the air, into a toroidal collector, and was distributed to each of the 14 nozzle-inlet passages. After the air had passed through the nozzles, it entered the rotor at the blade tips, passed through the rotor, and discharged into the depression tank in

which the turbine assembly was mounted. The air was then removed from the tank by the laboratory exhaust facilities. Turbine-inlet and -outlet pressures were regulated by butterfly throttle valves located upstream and downstream of the turbine.

Part of the inlet ducting and the depression tank was insulated to minimize heat transfer. An external oiling system with provision for heating the oil was used because the turbine-outlet temperatures were of the order of  $-140^{\circ}$  F.

The turbine-outlet inner shrouds were also insulated (fig. 2) to prevent the oil from freezing. The entire turbine air-flow passage was sealed and pressure-checked to prevent any possibility of air leakage.

### INSTRUMENTATION

The air-weight flow through the turbine was measured by the submerged adjustable orifice located in a straight section of the inlet ducting. Instrumentation of the turbine proper is shown in figure 2. Two total-pressure tubes, two static-pressure taps, and two thermocouples were located in each of the 14 nozzle-inlet passages immediately outside the tank bulkhead. This instrumentation was used to determine the state of the air entering the turbine. After the air had passed through the 14 inlet passages, it was turned through  $90^{\circ}$  vaned elbows (fig. 1(a)) and entered the nozzles. A total-pressure rake was installed downstream of the turning vanes in one of the nozzle passages to determine the extent of the total-pressure drop caused by the elbow and the turning vanes. Static-pressure taps were located on each side of a nozzle on a geometric mean-flow path along the length of the nozzle. Two static-pressure taps were also installed in the 2-inch radial-clearance space between the nozzles and the rotor and on both the front and rear turbine outer shrouds.

Outlet measurements were taken in the depression tank. Two thermocouple rakes on opposite sides of the tank together with two total-pressure rakes were used to determine the outlet state of the air. Each of the thermocouples and pressure tubes was located at the root-mean-square radius of three annular areas. Wall static taps were also located in the same plane along the circumference of the depression tank. All the depression-tank instrumentation was located well downstream of the turbine assembly.



The precision of the aforementioned measurements is estimated to be within the following limits:

Temperature, °R . . . . .	±1.0
Pressure, in. Hg . . . . .	±0.05
Air-weight flow, lb/sec . . . . .	±0.5
Turbine speed, rpm . . . . .	±10

#### PROCEDURE

The investigation was divided into the following three phases:

- (1) Turbine performance with varying rotor speed
- (2) Turbine performance for various degrees of partial admission
- (3) Turbine performance for various percentages of throttling at the turbine inlet

In order to establish turbine performance with various rotor speeds, the unit was operated at arbitrarily selected nominal equivalent rotor speeds of 2380, 3950, 5200, 6800, 8100, and 9400 rpm. For each speed, the inlet-air pressure and temperature were maintained at approximately 40.0 inches of mercury absolute and 468° R, respectively. For full admission, the maximum over-all pressure ratio over the entire speed range was limited to 3.2 by the capacity of the laboratory exhaust facilities for the particular apparatus used in this investigation. In order to establish turbine-performance trends with rotor speed at higher over-all pressure ratios, the unit was operated with three of the inlet passages closed at nominal equivalent rotor speeds of 5200, 6800, and 8100 rpm. With this configuration, the over-all pressure-ratio range was increased to 4.5 for the three speeds investigated.

The turbine performance for various degrees of partial admission was determined by maintaining nominal equivalent rotor speed, turbine-inlet pressure, and temperature at approximately 6800 rpm, 40.0 inches of mercury absolute, and 468° R, respectively. The unit was operated over a range of pressure ratios  $p'_2/p'_3$  from 1.8 to 7.5 for degrees of partial admission corresponding to 14 (full admission), 11, 8, 6, and 4 active nozzles. (As the number of active nozzles is decreased, the range of available pressure ratio is increased.)

The effectiveness of inlet throttling as a control parameter was ascertained by again maintaining the nominal equivalent rotor speed and inlet-air temperature at approximately 6800 rpm and 468° R, respectively, and operating the turbine over a range of pressure ratios  $p'_1/p'_3$  from



2.0 to 6.0 for several inlet pressures corresponding to percentages of inlet throttling of 0, 20, 35, 50, and 65. (The ratio  $p'_1/p'_3$  increases as the percentage of inlet throttling is increased.)

An appreciable total-pressure drop occurred through the vaned elbows of the nozzles (caused by the reversed air flow across the airfoil-shaped turning vanes, as shown in fig. 1(a)). This pressure drop was approximately 2.2 inches of mercury at the maximum weight flow. Because this condition could be eliminated in a permanent installation, all turbine-inlet pressures and pressure ratios were based on the total pressure measured by the total-pressure rake downstream of the turning vanes.

Periodic inspections of the unit were made during the investigation to observe any detrimental effects caused by the operation of the turbine at the low outlet temperatures.

## RESULTS AND DISCUSSION

Inasmuch as the principal objective of this investigation was to determine the effectiveness of the compressor when operated as a refrigeration turbine, the performance results are presented in a form that readily shows the refrigerating effect. This effect is described by the product of equivalent temperature drop  $\Delta T/\theta$ , equivalent air-weight flow  $w/\theta/\delta$ , and specific heat of air at constant pressure  $c_p$ . The performance curves show the equivalent air-weight flow and temperature drop for several turbine rotor speeds, pressure ratios, degrees of partial admission, and percentages of inlet throttling. The operation of the unit at low outlet temperatures and an observed erosion problem will be discussed.

### Turbine Performance at Various Rotor Speeds

The turbine performance for various nominal equivalent rotor speeds is presented in figure 3. The equivalent temperature drop and equivalent air-weight flow are shown for a range of equivalent speeds from 2380 to 9400 rpm and for a range of over-all pressure ratios  $p'_2/p'_3$  (total pressure entering the turbine nozzles divided by the total pressure in the depression tank) from 1.6 to 4.5. Adiabatic temperature-drop efficiencies are shown as contours on the plot.

With full admission, the maximum equivalent temperature drop obtained at a pressure ratio of 3.2 is  $114^\circ\text{R}$ , and occurs at an equivalent rotor speed between 6800 and 8100 rpm (fig. 3). The corresponding



equivalent air-weight flow is 24.0 pounds per second. When the pressure ratio across the turbine is progressively increased from 1.6 to 3.2, the maximum temperature drop occurs at increasing weight flows and rotor speeds. The maximum temperature drop at a given over-all pressure ratio represents the maximum adiabatic turbine efficiency for that pressure ratio. The refrigerating effect (or equivalent total heat extracted from the air/sec) is expressed by the equation

$$Q = \frac{w\sqrt{\theta}}{\delta} c_p \frac{\Delta T}{\theta}$$

The maximum refrigerating effect (or  $Q$ ) is indicated on figure 3. At a constant pressure ratio, the maximum refrigerating effect occurs at slightly higher values of equivalent air-weight flow than that at which the maximum equivalent temperature drop occurs. Just beyond these maximum values of temperature drop, the weight flow is increasing at a greater rate than the temperature drop is decreasing. Operation of the unit along this line is necessary in order to obtain the maximum possible refrigerating effect. This condition requires that the rotor speed be varied, however, thereby complicating the control and the operation of the unit.

By sacrificing some available refrigerating effect, a constant turbine operating speed may be chosen to simplify the control and the operation of the unit. The selection of this speed depends upon the available pressure ratio, turbine-inlet temperature, turbine-inlet pressure, and the requirements of the particular refrigeration problem. A nominal equivalent rotor speed of 6800 rpm was selected to satisfy the refrigeration demands for a particular application at this laboratory. At higher pressure ratios with full admission, this speed lies in the range of maximum efficiency and refrigerating effect. Operation of the unit at even higher pressure ratios with 11 active nozzles shows that even greater temperature drops were obtained. Reducing the number of active nozzles lowers the efficiency of the turbine so that with full admission a similar trend of increased temperature drop with increasing pressure ratio could be expected. Also the 6800-rpm turbine-speed curve is slightly to the right of the maximum values of equivalent temperature drop corresponding to a given over-all pressure ratio and indicates the turbine speed of 6800 rpm is reasonably close to maximum values of refrigerating effect at over-all pressure ratios up to 4.5. With turbine operating speed specified, it was necessary to determine a method of control that would allow a wide range of refrigerating effect. Two methods of control were investigated, partial admission and inlet throttling.



### Turbine Performance with Partial Admission

The variation of equivalent temperature drop with equivalent air-weight flow for 14 (full admission), 11, 8, 6, and 4 active nozzles at values of over-all pressure ratio from 1.8 to 7.5 and a nominal equivalent rotor speed of 6800 rpm is shown in figure 4(a). Adiabatic temperature-drop-efficiency contours are also presented. At the higher over-all pressure ratios, the loss in temperature drop incurred by blocking successive nozzles is relatively small until eight nozzles remain active. Losses incurred by operating with less than eight active nozzles are appreciably larger. For this turbine, efficient operation is therefore limited to a minimum of eight active nozzles. These results agree qualitatively with another turbine partial-admission investigation (reference 1).

The variation in refrigerating effect ( $Q$ ), air horsepower, and temperature drop with weight flow for several degrees of partial admission and various over-all pressure ratios is presented in figure 4(b). From this figure, it is apparent that by varying the pressure ratio and the number of active nozzles the refrigerating effect (or heat extraction) can be varied from 10 to 665 Btu per second, the weight flow from 4.8 to 24.7 pounds per second, and the temperature drop from  $9^{\circ}$  to  $142^{\circ}$  R.

The use of partial admission as a method of controlling the air-weight flow and the temperature drop of the turbine permits only step-wise operation corresponding to whole numbers of active nozzles. A mechanical system for achieving partial admission would be relatively simple to apply to the configuration of this particular experimental unit. It is possible, however, that other configurations would offer a more difficult mechanical problem.

### Turbine Performance with Inlet Throttling

Percentage of inlet throttling is hereinafter defined as the ratio of the total-pressure drop across the throttle to the throttle upstream total pressure  $(p'_1 - p'_2)/p'_1$  and over-all pressure ratio is defined by  $p'_1/p'_3$ .

The variation in equivalent temperature drop with equivalent air-weight flow for percentages of inlet throttling of 0, 20, 35, 50, and 65 and for over-all pressure ratios from 2.0 to 6.0 at a nominal equivalent rotor speed of 6800 rpm is shown in figure 5(a). By throttling at the turbine inlet, a range of temperature drop and weight flow may be obtained but only at the sacrifice of efficient turbine operation, as indicated by



the adiabatic temperature-drop-efficiency contours. For example, given an over-all pressure ratio of 2.4, a change in weight flow from 22.9 to 8.4 pounds per second results in a decrease in temperature drop of  $63.5^{\circ}$  R and a decrease in adiabatic efficiency from 78 to somewhat less than 30 percent. The corresponding change in temperature drop for partial admission (fig. 4(a)) is only  $48.5^{\circ}$  R, which represents a decrease in efficiency from 78 to slightly less than 50 percent. Partial admission is therefore the more efficient method of controlling the air-weight flow and temperature drop of the turbine.

The variation in refrigerating effect (Q), air horsepower, and temperature drop with weight flow for several percentages of inlet throttling and over-all pressure ratios is presented in figure 5(b) at a nominal equivalent rotor speed of 6800 rpm. From figure 5(b), it is apparent that by varying the percentage of inlet throttling and pressure ratio the refrigerating effect can be varied from 12 to 665 Btu per second, the weight flow from 4.4 to 24.7 pounds per second, and the temperature drop from  $10^{\circ}$  to  $113^{\circ}$  R. Although partial admission is the more efficient method of controlling the weight flow and the temperature drop through the turbine, inlet throttling does not require step-wise operation and presents no serious mechanical problem in its application to a unit.

#### Practical Aspect of Operation

Because the unit was submerged in an environment where the temperature was as low as  $-140^{\circ}$  F, some concern was felt for the mechanical operation of the unit. The modification of the oiling system by adding insulation and heat exchangers was adequate to keep the oil from freezing. Bearing and oil temperatures remained at approximately  $140^{\circ}$  F. Examination of the component parts of the unit after 80 hours of operation revealed that the only serious mechanical problem was that of erosion.

Periodic examination of the unit during the investigation revealed that a fine metallic powder was being continuously deposited on all surfaces inside the depression tank. Qualitative chemical analysis indicated strong traces of magnesium with small traces of aluminum and other metals. The initial rate of magnesium-powder formation was relatively high and appeared to decrease as the running time of the unit increased.

Photographs of the turbine rotor and the turbine outer shroud and nozzles are shown in figure 6. Erosion occurred on the surfaces that appear as light areas in these figures. These surfaces have the appearance of being sandblasted. The pressure surfaces of the nozzles were eroded to a greater extent than any of the other surfaces.



One possible explanation of this erosion phenomenon is that for most of the running time the inlet air was saturated at a pressure of 40 inches of mercury absolute and a temperature of 8° F. For a typical point with these inlet conditions and maximum weight flow, calculations, which were based on the measured static-pressure variation along the length of a nozzle and in the clearance space between the nozzles and the turbine rotor, indicate that approximately 95 percent of the water vapor present in the inlet air would have sublimated or condensed in the form of minute ice particles (calculated static temperature of the air entering the rotor was approximately -68° F) before the air entered the turbine rotor. Ice particles formed in the nozzles, obeying the laws of momentum, impinged upon the pressure surfaces of the nozzles causing them to erode. The eroded particles from the nozzles and the ice particles were swept by the incoming air stream to the rotor and there, because of their reduced velocity, bombarded the rear face of the rotor blades. Some of these particles were thrown back into the nozzles, intensifying the erosion already present and the remainder passed through the rotor, out of the unit, and into the depression tank. This reasoning does not explain why the rate of erosion decreased with running time; however, the same phenomenon of a decrease in erosion rate has been observed in the "wet" stages of steam turbines (reference 2).

If the premise is accepted that the erosion was principally centered in the nozzles, it would be possible to coat the nozzle surfaces with an erosion-resistant substance such as rubber and thus minimize the effects of erosion.

#### SUMMARY OF RESULTS

From an investigation to determine the feasibility of operating a centrifugal compressor as a radial centripetal refrigeration turbine, the following results were obtained:

1. Operation of a centrifugal compressor as a centripetal refrigeration turbine with reasonable efficiency was feasible.
2. A comparison of partial admission and inlet throttling as a method for control of the unit indicated that partial admission was the more efficient method, whereas inlet throttling was the simpler to apply.

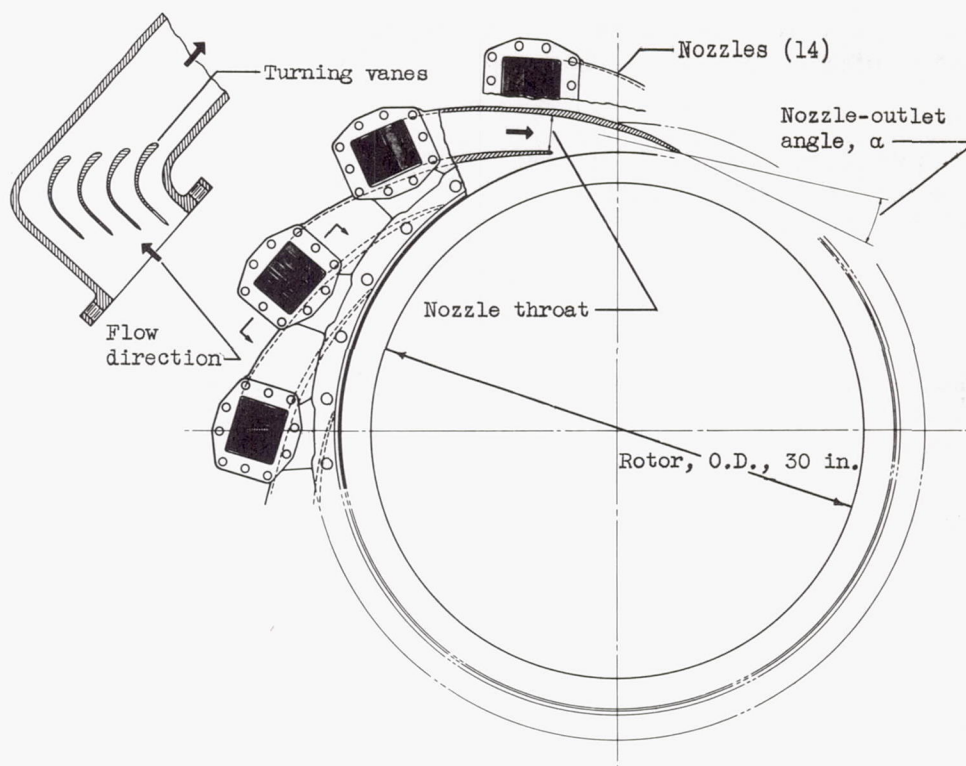


3. The nozzles should have erosion-resistant surfaces to minimize an erosion effect on the nozzles and the rotor.

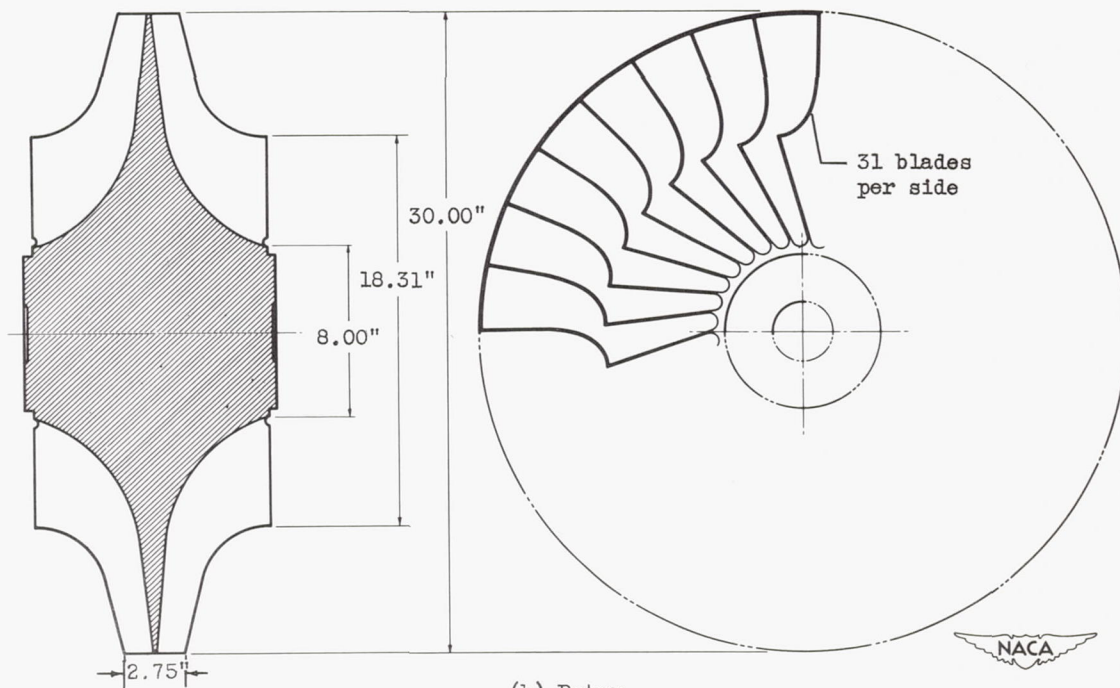
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#### REFERENCES

1. Kohl, Robert C., Herzig, Howard Z., and Whitney, Warren J.: Effects of Partial Admission on Performance of a Gas Turbine. NACA TN 1807, 1949.
2. Kraft, E. A.: The Modern Steam Turbine. VDI-Verlag G.M.B.H. (Berlin), 1931, p. 8.



(a) Nozzles.



(b) Rotor.

Figure 1. - Diagrammatic sketch of nozzles and rotor of refrigeration turbine.



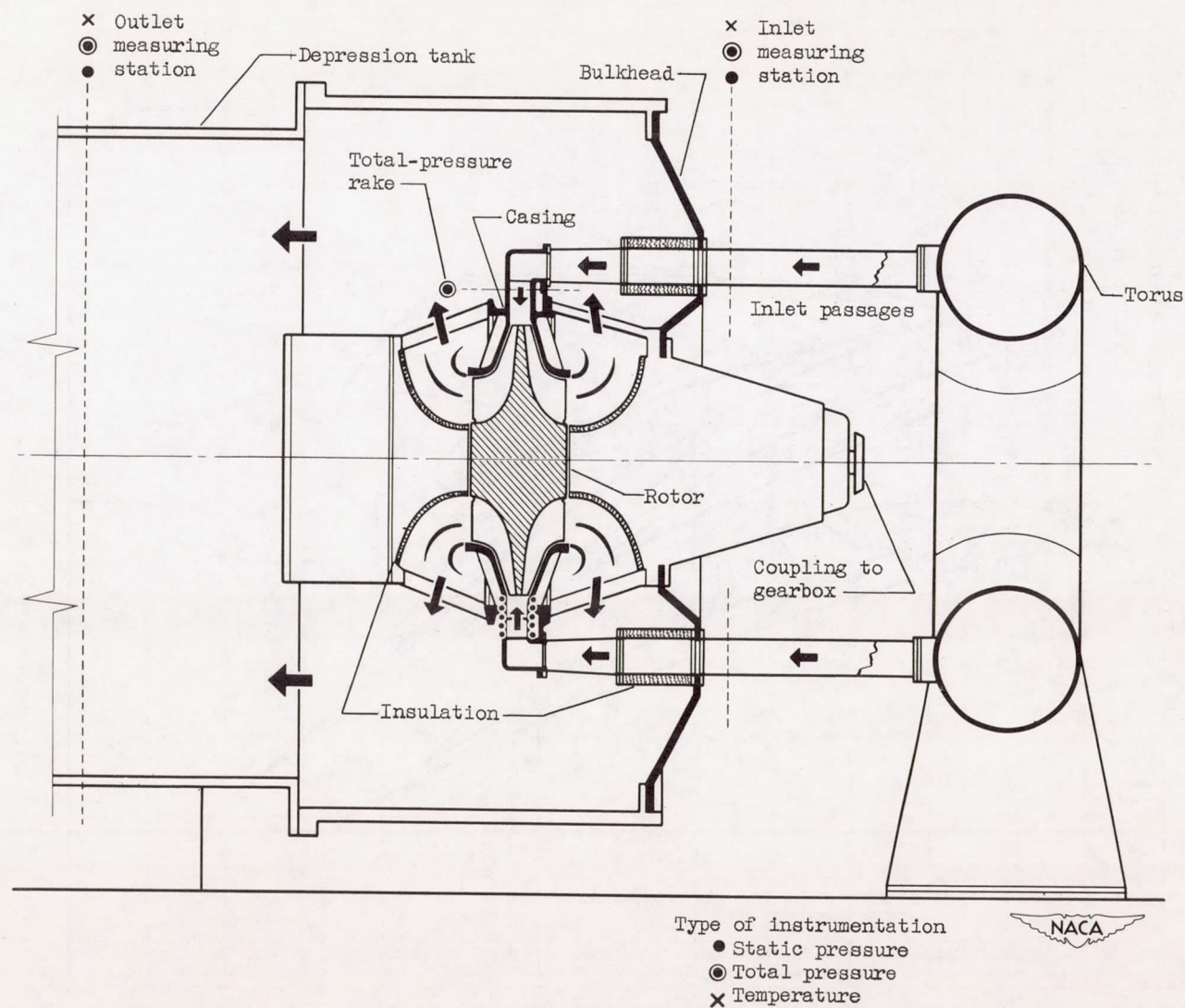


Figure 2. - Schematic diagram of refrigeration-turbine assembly and instrumentation.

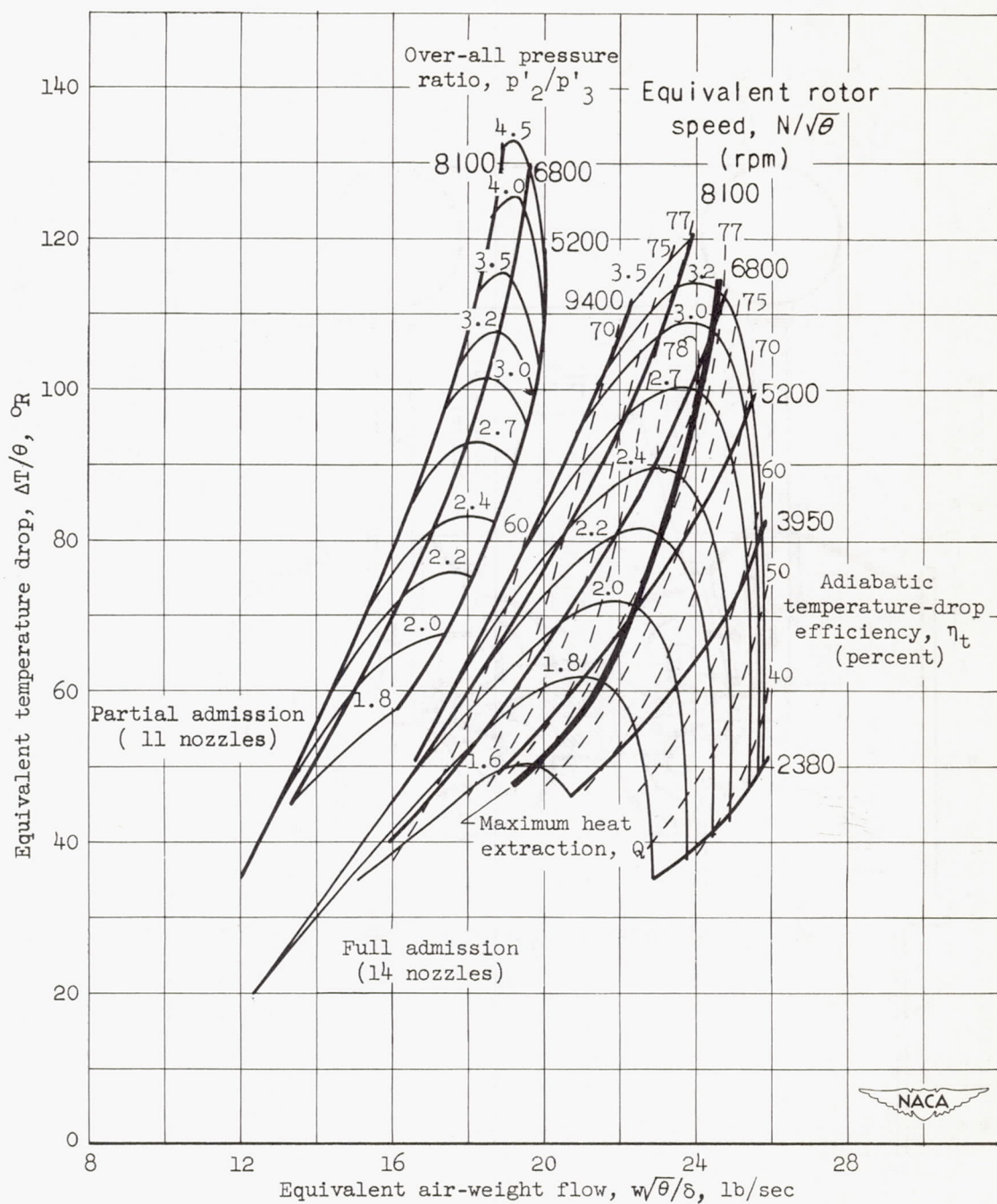
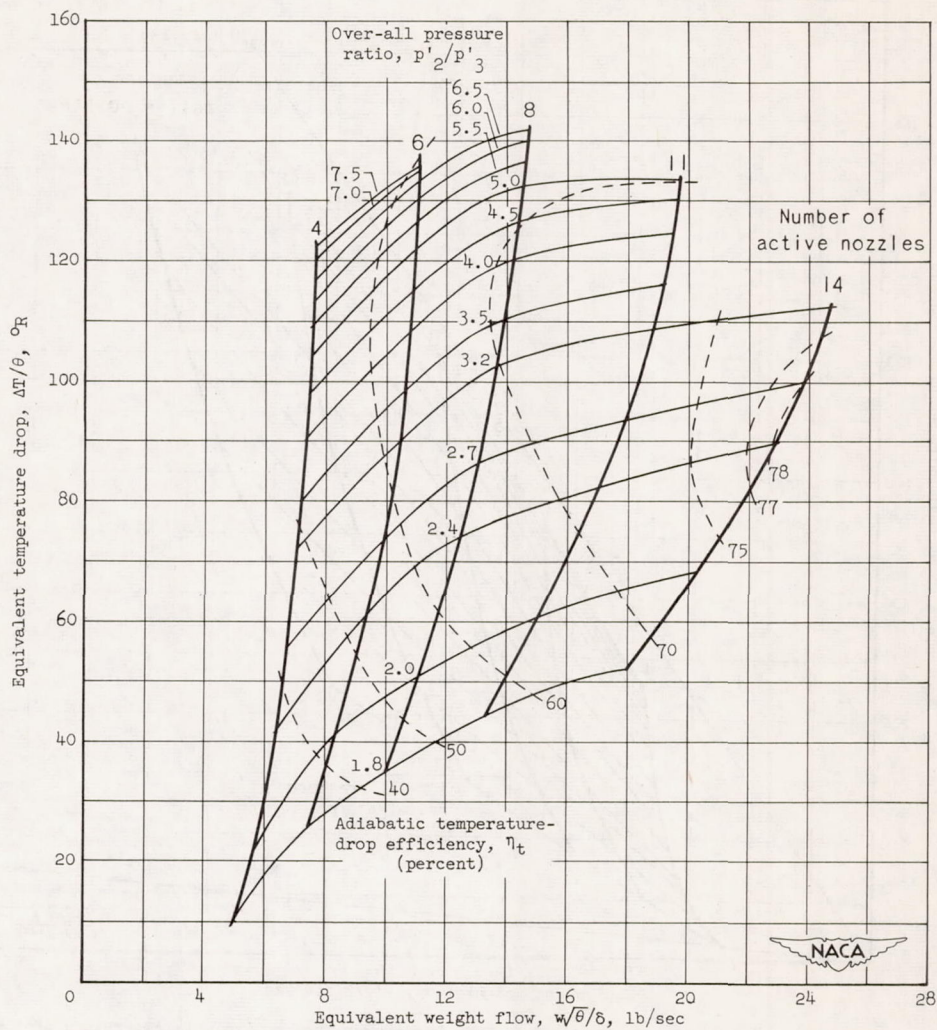


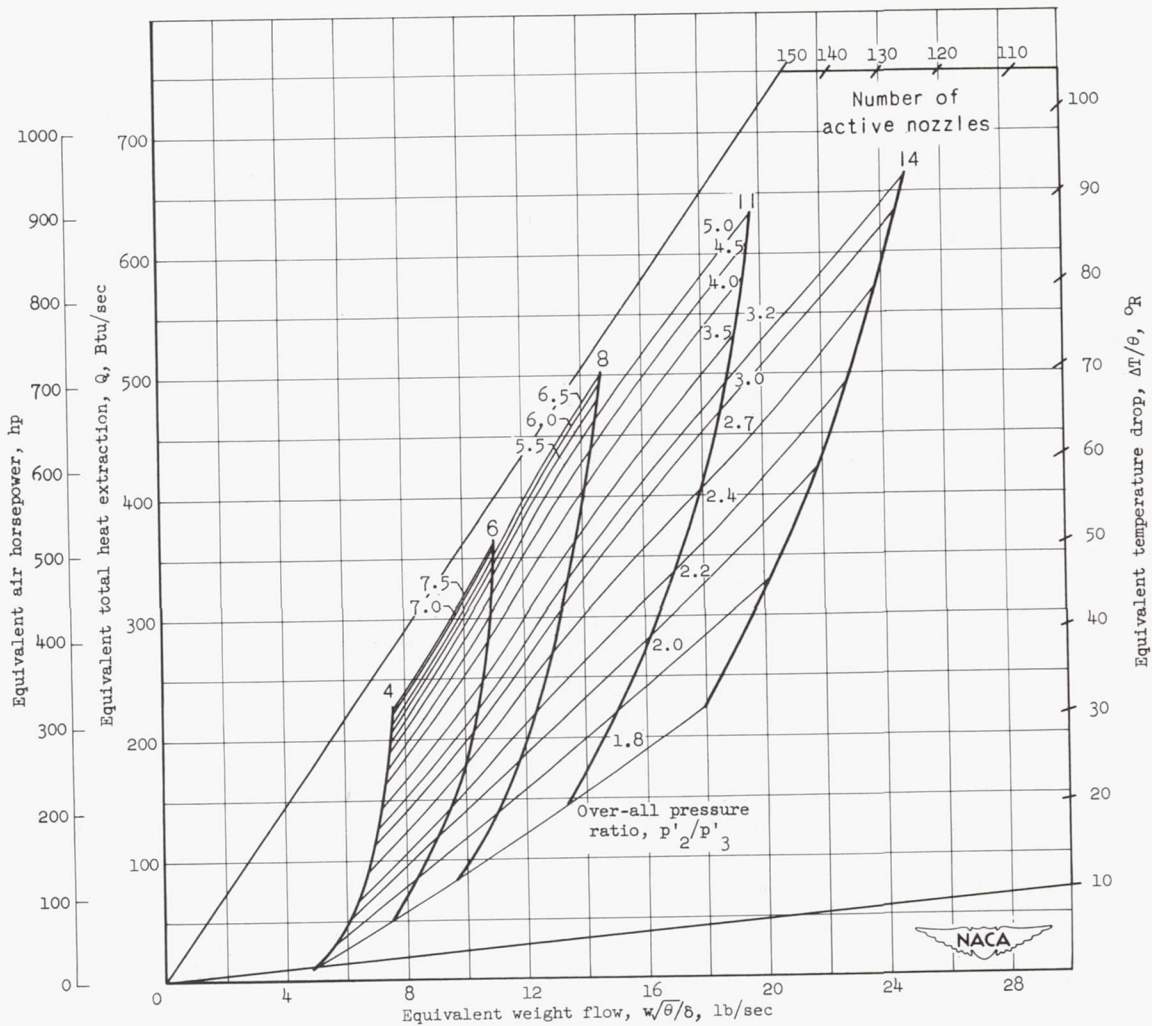
Figure 3. - Turbine performance at various nominal equivalent rotor speeds. Inlet-air temperature,  $468^\circ\text{R}$ ; inlet-air pressure, 40 inches mercury absolute.





(a) Variation of equivalent temperature drop with equivalent air-weight flow.

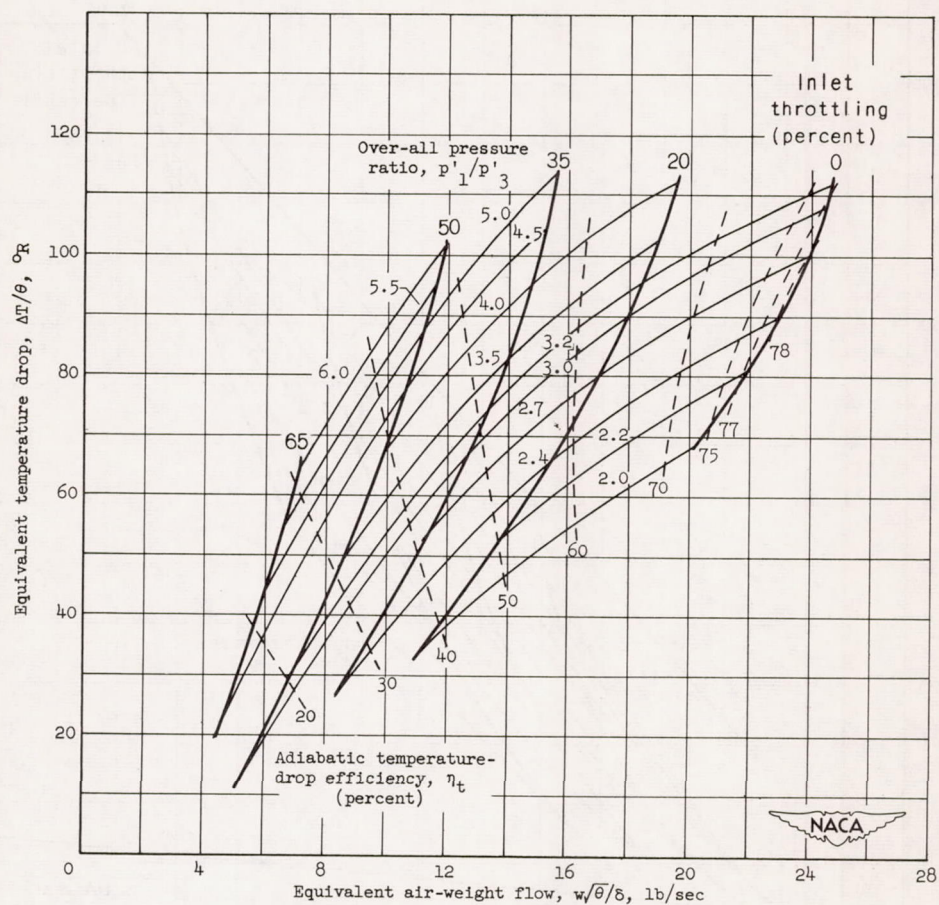
Figure 4. - Turbine performance at various degrees of partial admission and values of over-all pressure ratio. Equivalent rotor speed, 6800 rpm; inlet-air temperature,  $468^{\circ}\text{R}$ ; inlet-air pressure, 40 inches mercury absolute.



(b) Variation of refrigerating effect, air horsepower, and equivalent temperature drop with equivalent air-weight flow.

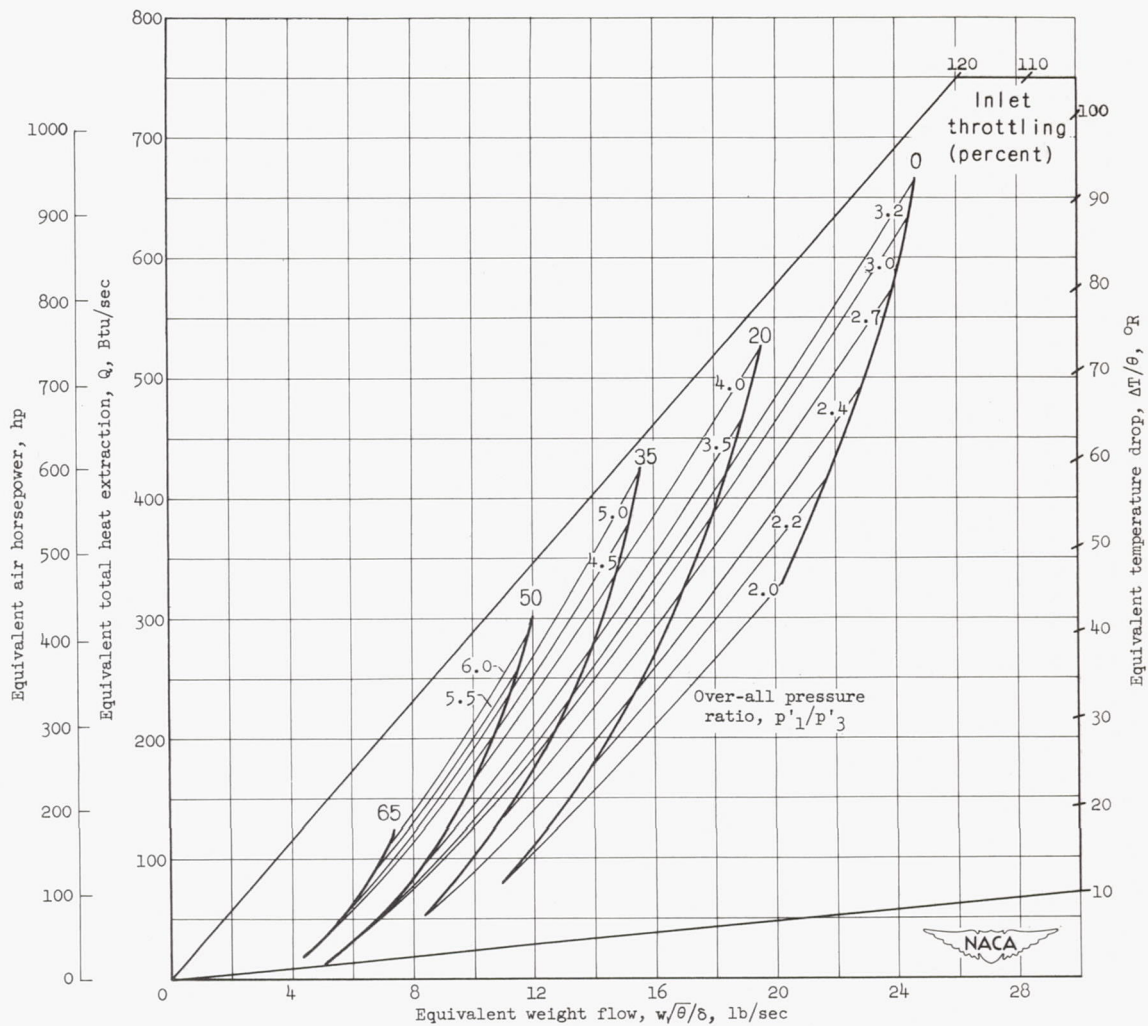
Figure 4.-Concluded. Turbine performance at various degrees of partial admission and values of over-all pressure ratio. Equivalent rotor speed, 6800 rpm; inlet-air temperature, 468° R; inlet-air pressure, 40 inches mercury absolute.





(a) Variation of equivalent temperature drop with equivalent air-weight flow.

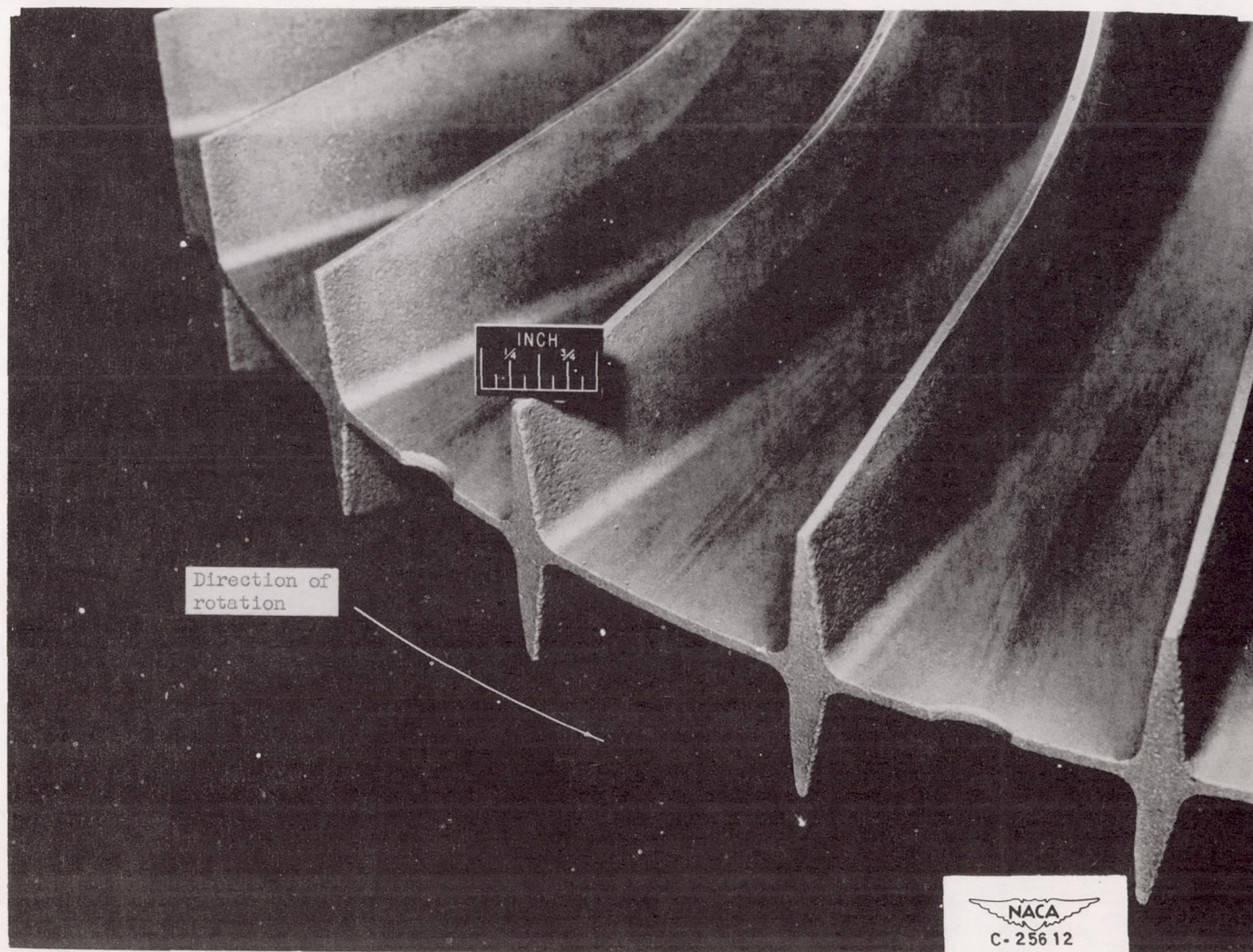
Figure 5. - Turbine performance at various percentages of inlet throttling and values of over-all pressure ratio. Equivalent rotor speed, 6800 rpm; inlet-air temperature, 468° R; throttle upstream pressure, 40 inches mercury absolute.



(b) Variation of refrigerating effect, air horsepower, and equivalent temperature drop with equivalent air-weight flow.

Figure 5. - Concluded. Turbine performance at various percentages of inlet throttling and values of over-all pressure ratio. Equivalent rotor speed, 6800 rpm; inlet-air temperature, 468° R; throttle upstream pressure, 40 inches mercury absolute.





(a) Rotor.

Figure 6. - Erosion of turbine.





